

# A non-tubular Stirling engine heater for a micro solar power unit

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## ABSTRACT

This paper presents a non-tubular heat exchanger for use in a Stirling solar engine micro co-generation unit that is being developed by the University of Oviedo and the technological research centre Tekniker Foundation. The engine has been designed using scaling criteria, which imply a case of relaxed dynamic similarity, that is to say, without preserving the equality among all of the prototype's and the derivative's dimensionless groups. The geometrical characteristics of the heater are described and the variables that can influence the pressure drop and heat transfer in the heater are identified. The correlations for the friction factor and Stanton number have been measured under steady flow conditions to analyse the feasibility of the experimental heater. The Reynolds analogy is not fulfilled, and the compressibility effects are negligible. The comparison between the correlation predictions for the non-tubular heater and slotted or tubular heaters of well-known prototypes shows possible applications for heaters that reach high Reynolds numbers and that encounter practical problems associated with the use of bundles of tubes.

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## 1. Introduction

Several countries are currently investigating distributed power generation systems fed by renewable energy to alleviate the saturation of supply networks in developed areas, as well as to provide an alternative for the electrification of rural districts in developing countries [1].

Dish-Stirling systems are currently a commercial reality [2], with the highest efficiency in solar conversion at power levels that are interesting for distributed power generation [3].

The engines in these systems have been adapted from prototypes originally designed for other applications. The heaters in those engines are variations of tubular heater heads located at the focal plane of a paraboloid solar concentrator.

The University of Oviedo and the technological research centre Tekniker Foundation, from Eibar, Spain, are engaged in a joint venture to develop a Stirling solar engine micro co-generation unit. The engine is currently being tested and is expected to serve low power applications, for which Stirling systems can be competitive [4].

The prototype has been designed to use two possible types of heaters, one of which has non-tubular geometry. The present paper focuses on the geometrical and physical features of a non-tubular Stirling engine heater, which could be interesting from both the design and performance viewpoints.

## 2. Characteristics of usual Stirling engine heaters

Philips' research has remarkably influenced the evolution of Stirling engine heater design. Since the fifties, the usual approach has consisted in slotted or tubular heaters supplied by combustion heating systems. Those tubes encountered problems related to their design, manufacturing and maintenance, such as the lack of thermal distribution uniformity, the risks of corrosion and overheating, the cost of special high temperature and high pressure resistant alloys, the combustion products fouling, etc. The tubes were usually joined to the cylinders by brazing either in hydrogen or vacuum atmosphere because of their inaccessibility for welding. A typical example is the helical heater-tube cage of the 4-215 engine tested in the Ford Torino around 1975. The tubes were made of Multimet with AISI-310 fins arranged on the outside branches. Around 1976, Meijer asserted that the complexity of the heater tubes made of exotic materials must be reduced to make them price-competitive [5].

However, substituting or modifying a typical tubular heater must preserve the flow symmetry as much as possible without penalising the dead volume or other parameters that influence the working fluid friction and heat transfer. The case of the SOLO V160 alpha-type co-generation engine is an example of an adaptation to solar applications. Since the nineties, SOLO has developed improved units with various heaters consisting of tubes brazed to tower manifolds. These heaters have accumulated thousands of operation hours with direct solar reception in Dish-Stirling units

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| Nomenclature |                                                                                                                               |                               |                                                                                             |
|--------------|-------------------------------------------------------------------------------------------------------------------------------|-------------------------------|---------------------------------------------------------------------------------------------|
| $A_{wxe}$    | wetted area of heater, m <sup>2</sup>                                                                                         | $R_{he}$                      | characteristic hydraulic radius of heater, m = $4V_{dxe}/A_{wxe}$                           |
| $A_{xxe}$    | cross-sectional area of heater, m <sup>2</sup>                                                                                | $t$                           | time, s                                                                                     |
| $c_p$        | specific heat of gas at constant pressure, J/(kg K)                                                                           | $T, T_g$                      | gas temperature, K                                                                          |
| $c_v$        | specific heat of gas at constant volume, J/(kg K)                                                                             | $T_w$                         | wall temperature, K                                                                         |
| $c_w$        | specific heat of wall material, J/(kg K)                                                                                      | $u$                           | gas velocity, m/s                                                                           |
| $C_f$        | instantaneous friction factor = $N_{eu}(R_{he}/L_e)$                                                                          | $\bar{u}$                     | mean gas velocity, m/s                                                                      |
| $h$          | convective heat transfer coefficient, W/(m <sup>2</sup> K)                                                                    | $V_{de}$                      | dead volume of expansion space, m <sup>3</sup>                                              |
| $k$          | thermal conductivity of the gas, W/(m K)                                                                                      | $V_{dxe}$                     | dead volume of heater, m <sup>3</sup>                                                       |
| $L_e$        | characteristic heater length, m                                                                                               | $V$                           | cylinder volume, m <sup>3</sup>                                                             |
| $\dot{m}$    | mass flow rate, kg/s                                                                                                          | $V_{sw}$                      | swept volume, m <sup>3</sup>                                                                |
| $n_s$        | engine frequency, rev/s                                                                                                       | $W_0$                         | quasi-static work per cycle, J                                                              |
| $N_{eu}$     | instantaneous Euler number<br>= $\Delta p / \left( \frac{1}{2} \rho \bar{u}^2 \right) = 2(\Delta p/p) N_{ma}^{-2}$            | $x$                           | longitudinal coordinate along the heater length, m                                          |
| $N_{fo}$     | instantaneous, spatially-averaged Fourier number = $\alpha_w / R_{he} \sqrt{RT_{ref} \cdot 1/N_{ma}}$                         | $\alpha_{xxe}$                | cross-sectional area ratio = $A_{xxe}/V_{sw}^{2/3}$                                         |
| $N_{ma}$     | instantaneous, spatially-averaged, isothermal Mach number = $\bar{u} / \sqrt{RT_{ref}} = \dot{m} \sqrt{RT_{ref}} / p A_{xxe}$ | $\alpha_w$                    | thermal diffusivity of wall material, m <sup>2</sup> /s                                     |
| $N_{MA}$     | characteristic Mach number = $n_s V_{sw}^{1/3} / \sqrt{RT_C}$                                                                 | $\alpha_{wxe}$                | wetted area ratio = $A_{wxe}/V_{sw}^{2/3}$                                                  |
| $N_p$        | characteristic pressure number = $N_{SG} N_{MA}$                                                                              | $\beta$                       | drag coefficient in Forchheimer's law, m <sup>-1</sup>                                      |
| $N_{pr}$     | Prandtl number = $\mu c_p / k$                                                                                                | $\lambda_1, \dots, \lambda_k$ | dimensionless geometrical parameters, including those characteristic of the drive mechanism |
| $N_{re}$     | instantaneous, spatially-averaged Reynolds number = $4 \rho \bar{u} R_{he} / \mu = 4 \dot{m} R_{he} / \mu A_{xxe}$            | $\Phi$                        | coefficient of indicated power losses                                                       |
| $N_{sg}$     | instantaneous, spatially-averaged Stirling number = $N_{ma}^2 / N_{re}$                                                       | $\gamma$                      | adiabatic coefficient                                                                       |
| $N_{SG}$     | characteristic Stirling number = $p_m / \mu n_s$                                                                              | $\kappa$                      | swept volume ratio = $V_C / V_E$                                                            |
| $N_{st}$     | instantaneous, spatially-averaged Stanton number = $h / \rho \bar{u} c_p = h A_{xxe} / \dot{m} c_p$                           | $\kappa$                      | permeability coefficient in Forchheimer's law, m <sup>2</sup>                               |
| $N_{TCR}$    | instantaneous thermal capacity ratio = $\rho_w c_w / \rho c_p = \rho_w c_w T_{ref} / p \cdot (\gamma - 1) / \gamma$           | $\lambda_{he}$                | hydraulic radius ratio = $r_{he} / V_{sw}^{1/3}$                                            |
| $p$          | pressure, Pa                                                                                                                  | $\mu$                         | viscosity, Pa s                                                                             |
| $p_m$        | mean pressure, Pa                                                                                                             | $\mu_{de}$                    | dimensionless dead volume of expansion space = $V_{de} / V_{sw}$                            |
| $P_B$        | brake power, W                                                                                                                | $\mu_{dxe}$                   | dimensionless dead volume of heater = $V_{dxe} / V_{sw}$                                    |
| $P_{ind}$    | indicated power, W                                                                                                            | $\Psi$                        | coefficient of indicated power losses                                                       |
| $R$          | specific gas constant, J/(kg K)                                                                                               | $\rho$                        | density of gas, kg/m <sup>3</sup>                                                           |
| $r_{he}$     | local hydraulic radius of heater, m                                                                                           | $\rho_w$                      | density of wall material, kg/m <sup>3</sup>                                                 |
|              |                                                                                                                               | $\tau$                        | temperature ratio = $T_C / T_E$                                                             |
|              |                                                                                                                               | $\zeta_{ind}$                 | dimensionless indicated power = $P_{ind} / (p_m V_{sw} n_s)$                                |
|              |                                                                                                                               | $\zeta_0$                     | quasi-static dimensionless work per cycle = $W_0 / (p_m V_{sw})$                            |
|              |                                                                                                                               | <b>Subscripts</b>             |                                                                                             |
|              |                                                                                                                               | $c, C$                        | compression space                                                                           |
|              |                                                                                                                               | $e, E$                        | expansion space                                                                             |

[6]. However, tower manifolds penalise dead volume and flow symmetry, which suggests that other approaches might increase both the reliability and the high efficiency of the SOLO units.

The flow symmetry is more adequate in the heater cage designed by Mahkamov et al. [7] for a 3 kW Stirling engine, which combines natural gas and solar energy in remote sites of Central Asia. The combustion chamber provides a cavity that surrounds the heater and has an input window at its top for the concentrated solar irradiation. The heater cage consists of two collectors and two sets of semi-circular rings made from stainless steel tubes. A different approach uses a direct solar heater-tube receiver placed at the bottom of a cylindrical cavity in the multi-cylinder United Stirling 4-95 engine, which holds the solar-electric efficiency world record [8].

To reduce the geometrical limitations when adapting heater-tube cages to solar conversion, heat-pipe receivers have been suggested as intermediate heat transfer systems. The STM4-120 prototype adapted by Stirling Thermal Motors is an example technology that allocates a solar dish heat-pipe receiver in multi-cylinder engines at high power ranges. At a low power range, the NALSEM 500 unit, originally conceived to provide solar power for space stations, uses an interface of sodium heat pipes to uniformly transfer heat from a cylindrical solar cavity receiver to the heater

head [9]. However, intermediate heat transfer systems have disadvantages for both the system's efficiency and costs, particularly for low power, terrestrial applications.

Additionally, despite the fact that the most efficient solar Stirling engines have been developed by Philips licensed companies, the design methods, comprehensive specifications and test results related to their prototypes have not been published. Independent authors, like Tew et al. [10], Tsuchiya et al. [11], Rice and Hennes [12], Lee et al. [13], and Organ et al. [14], have studied the gas dynamics and heat transfer of tubular or slotted Stirling heaters, but there are no current friction factors or Stanton or Nusselt number correlations that are corroborated enough for not-fully-developed, bidirectional flow with a variable mass rate.

The publications related to other geometries may be considered exceptional and incomplete. For example, Isshiki et al. [15] showed experimental results for an engine with a tubeless heater, but the heater performance characteristics were not reported.

In short, the Stirling engine heaters designed for solar applications usually have tubular arrangements whose development level does not seem definitive; meanwhile, innovations are hindered by the lack of knowledge about the performance of new geometrical concepts.

### 3. Geometrical description of the experimental heater

Similarity criteria previously published have been applied to the preliminary design of the Stirling engine developed by the University of Oviedo and the Tekniker Foundation [16–21]. The engine was scaled from the Philips M102C beta prototype [22], using air as the working fluid and a ‘bell’-crank mechanism. The mean pressure and engine speed were assumed to have more influence on the engine’s cost than the swept volume. Thus, a length scale factor of 1.6 was assumed, which should lead to a more powerful, slower and less pressurised engine using the same working fluid [23,24]. Table 1 shows the main characteristics and operating points for both the prototype and the derivative, assuming strict dynamic similarity, that is to say, the equality between all the dimensionless groups of the prototype and the derivative.

Scaling-based design methods provide criteria to adapt an engine for an application different from its original one. However, heater innovation is difficult without using out-to-scale geometrical parameters. Thus, the concept of ‘relaxed dynamic similarity’, introduced by Organ [16,17], should be taken into account. This concept produces an interesting derivative despite not maintaining the equality between all of the dimensionless groups of the prototype and the derivative. The practical rules for applying this procedure have been published [24]. For example, a model might be interesting with a heater whose hydraulic radius-to-length ratio is higher than the prototype value, which means the lack of geometric similarity, provided that the approximate equality among the rest of dimensionless groups, leads to an acceptable power.

The strict similarity was only maintained for the cylinder diameters, the piston strokes, and the geometrical characteristics of the slotted cooler, as well as for the wire diameter, porosity, free flow area and length of the regenerator because the compactness and low speed operation may justify the target of an alpha-type derivative with the Ross yoke mechanism. The combination of these design criteria led to an out-to-scale swept volume of  $V_{sw} = 341.82$  cc, which implies a case of relaxed similarity. Thus, the validity of the scaling procedure can only be accepted if the lack of strict similarity in several components does not lead to characteristic dimensionless parameters that would be unfavourable for the derivative operation.

Fig. 1 shows a schematic section of the experimental engine. The buffer space is enclosed at the bottom by a casing, which has been removed in the figure to show the drive mechanism and the electrical generator. The heater was manufactured using die-sinking electrical discharge machining (EDM) to form the required shape in the flat disc of stainless steel (shown at the top level of the figure) with a three-dimensional electrode of graphite.

The heater geometry consists of a flat surface intended to receive direct solar irradiation from a conventional optical concentrator, for example, a parabolic dish or a Fresnel lens-based system, that will then transmit it to the bases of almost one thousand 10-mm long and 2-mm diameter rods, around which the working gas circulates. The rods are arranged in a staggered manner with a uniform diagonal pitch on an oval area perpendicular to the cylinder axes, with ends

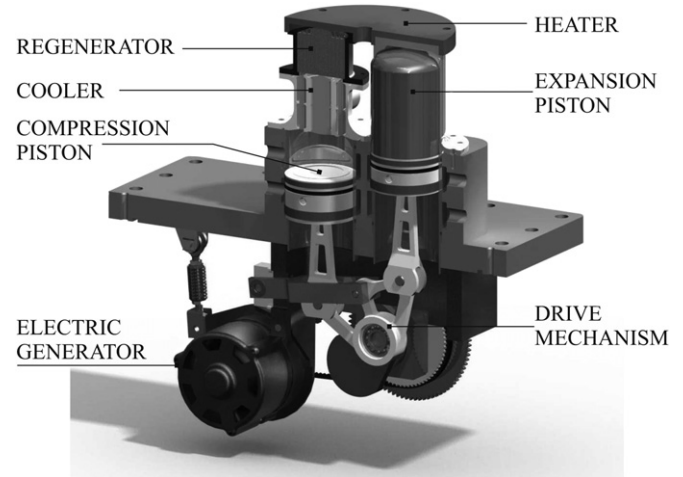


Fig. 1. A picture of the experimental engine.

that are connected to the hot cylinder and the regenerator casing, as shown in Figs. 2 and 3.

Table 2 allows the geometrical characteristics of the experimental heater to be compared with those of the prototype heater and the ones of a derivative designed under strict similarity conditions. The geometry of the experimental heater requires a particular definition for the heater length  $L_e$ , which has been defined as the distance between longitudinal axes of the cylinders. Thus,  $L_e$  is a specimen of ‘equivalent’ length that must be taken into account to accurately interpret the meaning of the cross-sectional area  $A_{xxe}$  and those of the non-dimensional ratios  $R_{he}/L_e$  and  $\alpha_{xxe}$ .

Dynamic similarity requires that the Stirling engine heater prototype and its derivative have the same values of a set of dimensionless parameters, namely, three independent values related to geometry (for example,  $\mu_{dxe}$ ,  $\alpha_{wx}$  and  $\lambda_{he}$ ) and those related to the working fluid and operating conditions ( $\tau$ ,  $\gamma$ ,  $N_p$  and  $N_{MA}$ ).

The distinction between the scope of Thermodynamics and other physical theories related to Gas Dynamics and Heat Transfer [25] is important when dimensional analysis techniques are applied. Such a distinction refers to the selection of both the dimensional basis and the physical quantities that influence the problem. The thermodynamic similarity is influenced by the temperature, mean pressure and volume parameters; therefore, the quasi-static dimensionless work per cycle may be expressed as follows [19,24]:

$$\zeta_0 = f(\tau, \kappa, \lambda_1, \dots, \lambda_k, \mu_{dce}, \mu_{de}, \mu_{dr}, \mu_{dc}, \mu_{dcc})$$

To take into account the features related to Gas Dynamics and Heat Transfer, i.e., the friction factor and Stanton or Nusselt numbers, the rest of the geometrical parameters, physical properties of both the working fluid and the regenerator material, and operating engine speed, must be added to the list of influencing quantities, which allows the following expression to be obtained for the dimensionless indicated power [19,24]:

$$\zeta_{ind} = f(\tau, \kappa, \lambda_1, \dots, \lambda_k, \mu_{dce}, \mu_{de}, \mu_{dr}, \mu_{dc}, \mu_{dcc}, \alpha_{xe}, \alpha_{xr}, \alpha_{xc}, \lambda_{hr}, \lambda_{hc}, \gamma, N_\alpha, N_{TCR}, N_p, N_{MA})$$

Prieto et al. [20] have proposed the following approach that explicitly states the influence of the engine speed on the dimensionless indicated power of kinematic Stirling engines:

Table 1

The main design parameters of the prototype and the derivative under strict similarity conditions.

|            | $T_E$<br>(°C) | $T_C$<br>(°C) | $V_E$<br>(cm <sup>3</sup> ) | $V_{sw}$<br>(cm <sup>3</sup> ) | $p_m$<br>(MPa) | $n_s$<br>(rpm) | $P_{ind}$<br>(W) | $P_B$<br>(W) |
|------------|---------------|---------------|-----------------------------|--------------------------------|----------------|----------------|------------------|--------------|
| M102C      | 600           | 60            | 59.4                        | 64.2                           | 1.103          | ~1860          | ~375             | ~163         |
| Derivative | 600           | 60            | 243.3                       | 262.8                          | 0.69           | ~1160          | ~600             | ~260         |

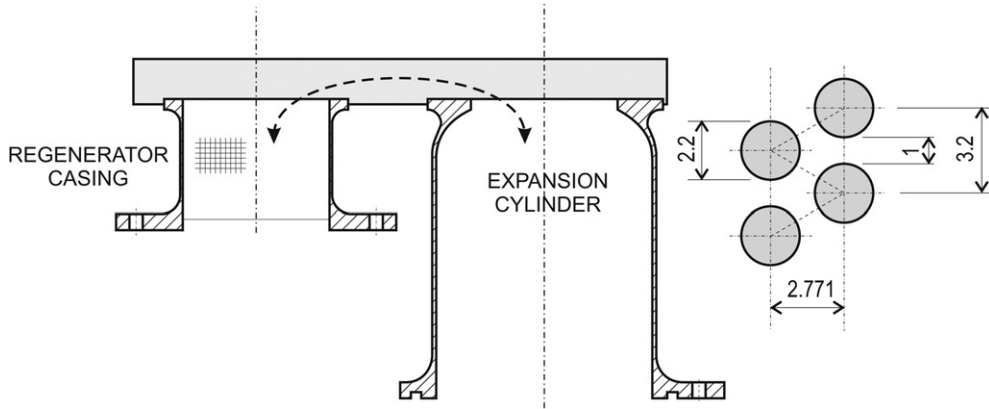


Fig. 2. A conceptual sketch of the new heater.

$$\zeta_{ind} = \zeta_0 - \Phi N_{MA} - \Psi N_{MA}^2 \quad (1)$$

Thus, in this equation,  $\zeta_0$  is a thermodynamic concept that represents the theoretical prediction, assuming an ideal cycle without any losses caused by leakage, thermal or mechanical irreversibilities, while the factors  $\Phi$  and  $\Psi$  of indicated power losses characterise irreversibilities inherent to working gas friction and heat transfer.

The new heater's dead volume is more than twice the value of the strictly similar heater based on thermodynamic similarity. This disadvantage has been compensated for by reducing the dead volume in both transition spaces adjacent to the expansion cylinder and the regenerator. Therefore, the value  $\zeta_0 \approx 0.302$  was computed by quasi-static simulation for the relaxed derivative, which is slightly higher than the value  $\zeta_0 \approx 0.294$  obtained for the prototype, with a different engine configuration and drive mechanism.

Table 2 shows that the experimental heater ratio  $R_{he}/L_e$  is more favourable with regards to the pressure losses caused by the working gas friction, at 41% higher than the strict derivative value. Conversely, the heat exchanging capability of the experimental heater might be compromised because its wetted area ratio  $\alpha_{wxe}$  is 19% lower than the strict derivative's value. This handicap is caused by the high swept volume derived from the new engine configuration and drive mechanism, despite the fact that the wetted surface is almost identical for both cases.

Thus, the geometrical design of the engine seems adequate based on the quasi-static performance; however, before accepting the feasibility of the new heater, additional analyses are required to evaluate the influence of non-geometrical variables on indicated power losses.

#### 4. Friction and heat transfer characteristics of the experimental heater

Assuming a one-dimensional model, the local instantaneous pressure drop through an elemental length of the experimental heater can be expressed as a function of: the pressure itself, the geometrical parameters, the fluid physical properties and the fluid velocity. To define the geometry for the purposes of this paper, the complete set of geometrical data related to the heater oval shape, diameter, length and rod arrangement, as well as other minor geometrical details, may be approximately substituted by the local hydraulic radius. The density and the gas viscosity are included in the list of variables to take into account in calculating the inertia or form drag and the viscous effects, respectively. The pressure is included to consider compressibility effects that may be present at high velocities. Thus, the following functional relationship can be applied:

$$\frac{\delta p}{\delta x} = f(p, r_{he}, \rho, \mu, u) \quad (2)$$

The integration of this function along the heater length leads to the instantaneous pressure drop at the heater  $\Delta p$ . The process allows the gas flow to be characterised using an instantaneous, spatially-averaged velocity defined as  $\bar{u} = \dot{m}/(\rho A_{xhe})$ . Moreover, the list of influential parameters is completed with the gas temperatures at both the entry and exit sections,  $T_{g0}$  and  $T_{gL}$ , as well as with the wall temperature, which usually is assumed to be uniform, to consider the spatial variations of gas properties. Alternatively,  $T_{g0}$  and  $T_{gL}$  can be replaced by the gas temperature difference  $\Delta T_g$  and the mean gas temperature  $\bar{T}_g$ . Therefore, it can be written:

$$\Delta p = f(p, R_{he}, \rho, \mu, \bar{u}, L_e, \Delta T_g, \bar{T}_g, T_w) \quad (3)$$

The application of Buckingham's theorem to the last expression (3) leads to the following functional relationship with six dimensionless groups, in which  $R_{he}$ ,  $\rho$ ,  $\bar{u}$  and  $T_w$  have been used as reference quantities:

$$\frac{\Delta p}{\frac{1}{2}\rho\bar{u}^2} = F\left(\frac{\mu}{\rho\bar{u}R_{he}}, \frac{p}{\frac{1}{2}\rho\bar{u}^2}, \frac{L_e}{R_{he}}, \frac{\Delta T_g}{T_w}, \frac{\bar{T}_g}{T_w}\right)$$

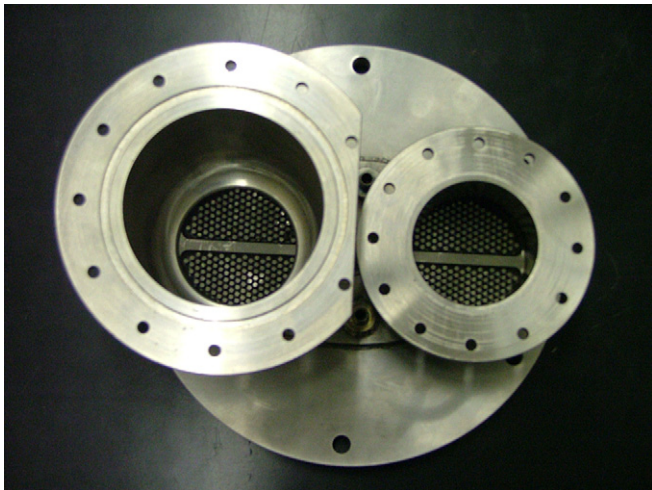


Fig. 3. The rods of the experimental heater at the ends of the hot cylinder and the regenerator casing.



**Table 2**

A comparison among the geometrical characteristic parameters for the prototype heater, the strictly similar derivative and the experimental heater.

|                    | $V_{dxe}$ (cm <sup>3</sup> ) | $V_{de}$ (cm <sup>3</sup> ) | $A_{wxe}$ (cm <sup>2</sup> ) | $R_{he}$ (mm) | $L_e$ (mm) | $A_{xxe}$ (cm <sup>2</sup> ) | $\mu_{dxe}$ (–) | $\mu_{de}$ (–) | $\alpha_{wxe}$ (–) | $\lambda_{he}$ (–) | $\alpha_{xxe}$ (–) | $R_{he}/L_e$ (–) |
|--------------------|------------------------------|-----------------------------|------------------------------|---------------|------------|------------------------------|-----------------|----------------|--------------------|--------------------|--------------------|------------------|
| M102C              | 5.67                         | 16.77                       | 331.38                       | 0.171         | 37.25      | 1.52                         | 0.088           | 0.261          | 20.679             | 0.0043             | 0.0949             | 0.0046           |
| Strict derivative  | 23.22                        | 68.69                       | 848.17                       | 0.274         | 59.60      | 3.89                         | 0.088           | 0.261          | 20.679             | 0.0043             | 0.0949             | 0.0046           |
| Relaxed derivative | 57.00                        | 76.85                       | 817.00                       | 0.698         | 108.00     | 5.28                         | 0.167           | 0.225          | 16.712             | 0.0100             | 0.1080             | 0.0065           |

The ideal gas model allows the pressure to be substituted by the product  $\rho RT_{ref}$ , where  $T_{ref}$  denotes the reference temperature at which the fluid properties are computed. Therefore, the pressure drop factor, sometimes called the Euler number, is a function of the Reynolds number, the Mach number, and the geometrical and temperature ratios:

$$N_{eu} = F\left(N_{re}, N_{ma}, \frac{L_e}{R_{he}}, \frac{\Delta T_g}{T_w}, \frac{\bar{T}_g}{T_w}\right) \quad (4)$$

The advantages of using the Stirling number to characterise the Stirling engine phenomena have been evidenced previously [21,26,27], but it is convenient to maintain the more common expression  $N_{re} = N_{ma}/N_{sg}$  in the last functional relationship to facilitate comparisons.

A merely algebraic substitution allows the Eq. (4) to be modified to introduce the fractional pressure drop as follows:

$$\frac{\Delta p}{p} = N_{ma}^2 \cdot F\left(N_{re}, N_{ma}, \frac{L_e}{R_{he}}, \frac{\Delta T_g}{T_w}, \frac{\bar{T}_g}{T_w}\right) \quad (5)$$

An analogous procedure can be applied to analyse the instantaneous convective heat transfer coefficient, which then requires that the list of independent variables of the Eq. (3) be completed with the addition of characteristic parameters of the gas thermal performance, namely, its specific heat capacities. Moreover, if the heat conduction in the heater rods is considered to influence the behaviour, the volumetric, specific heat capacity and the thermal diffusivity of the wall material must also be taken into account. The pressure is not explicitly included here because the product  $RT_{ref}$  is determined by the gas temperatures and the specific heat capacities. Thus, the following expression can be written:

$$h = f(R_{he}, \rho, \mu, \bar{u}, k, c_p, c_v, L_e, \Delta T_g, \bar{T}_g, T_w, \rho_w c_w, \alpha_w) \quad (6)$$

Applying Buckingham's theorem to the last expression leads to the following functional relationship that involves ten dimensionless groups, in which  $R_{he}$ ,  $\rho$ ,  $\bar{u}$  and  $c_p$  have been used as reference quantities:

$$\frac{h}{\rho \bar{u} c_p} = F\left(\frac{\mu}{\rho \bar{u} R_{he}}, \frac{L_e}{R_{he}}, \frac{k}{\rho \bar{u} c_p R_{he}}, \frac{c_v}{c_p}, \frac{c_p \Delta T_g}{\bar{u}^2}, \frac{c_p \bar{T}_g}{\bar{u}^2}, \frac{c_p T_w}{\bar{u}^2}, \frac{\rho_w c_w}{\rho c_p}, \frac{\alpha_w}{\bar{u} R_{he}}\right)$$

Taking into account that  $c_p/R = \gamma/(\gamma - 1)$  and manipulating the equation to obtain more meaningful dimensionless groups, this equation can then be written as follows:

$$N_{st} = F\left(N_{re}, \frac{L_e}{R_{he}}, N_{pr}, \gamma, \frac{\Delta T_g}{T_w}, \frac{\bar{T}_g}{T_{ref}}, N_{ma}^{-2}, \frac{\bar{T}_g}{T_w}, N_{TCR}, N_{fo}\right)$$

Thus, for  $T_{ref} = \bar{T}_g$ , the previous equation is finally reduced to:

$$N_{st} = F\left(N_{re}, N_{pr}, N_{ma}, \gamma, \frac{L_e}{R_{he}}, \frac{\Delta T_g}{T_w}, \frac{\bar{T}_g}{T_w}, \frac{\rho_w c_w \Delta T_g}{p}, \frac{\alpha_w}{R_{he} \sqrt{R \Delta T_g}}\right) \quad (7)$$

Eqs. (4), (5) and (7) are the basis of the experimental analysis described below. To correctly interpret the results, it must be noted that these equations can be applied to a variety of physical

phenomena whose understanding depends on how the same dimensionless numbers are interrelated.

For example, assuming isothermal conditions, Eqs. (4) and (5) reduce to the following well-known expressions that apply for the steady, fully-developed, laminar flow of incompressible, Newtonian fluids through smooth, straight ducts:

$$N_{eu} = \frac{16}{N_{re}} \cdot \frac{L_e}{R_{he}} \leftrightarrow \frac{\Delta p}{p} = N_{ma}^2 \cdot \frac{8}{N_{re}} \cdot \frac{L_e}{R_{he}} \leftrightarrow C_f = \frac{16}{N_{re}}$$

However, it can be noted that the introduction of Euler, Reynolds and Mach numbers in this problem is arbitrary because inertia, form drag and compressibility effects have not been considered. In other words, density and pressure are not influencing properties based on Hagen-Poiseuille's law:

$$-\frac{\delta p}{\delta x} = \frac{2\mu \bar{u}}{r_{he}^2} \approx \text{constant} \leftrightarrow \Delta p = \frac{2\mu \bar{u}}{R_{he}^2} L_e$$

The revision of the analysis after removing  $\rho$  and  $p$  from the list of variables shows that, in this case, only one dimensionless group is linked to the length-to-hydraulic radius ratio. That dimensionless group may be interpreted as a version of the Stirling number, which is the product of several of the abovementioned groups that, therefore, cannot vary independently:

$$\begin{aligned} \frac{\Delta p R_{he}}{\mu \bar{u}} &= 2 \frac{L_e}{R_{he}} \leftrightarrow N_{eu} \cdot N_{re} = 16 \frac{L_e}{R_{he}} \leftrightarrow \frac{\Delta p}{p} \cdot N_{re} \cdot N_{ma}^{-2} \\ &= 8 \frac{L_e}{R_{he}} \end{aligned}$$

Eqs. (4) and (5) can also be reduced to expressions that apply for cases where the density is needed for modelling. For example, Forchheimer's law takes into account the pressure drop not only caused by viscosity but also by the acceleration and deceleration effects of the fluid as it travels through the tortuous flow path of the porous media:

$$-\frac{\delta p}{\delta x} = \mu \frac{u}{\kappa} + \beta \rho u^2 \leftrightarrow \frac{(-\delta p / \delta x) / \beta}{\rho u^2} = \frac{\mu}{\rho u \kappa \beta} + 1$$

The spatial integration of this law allows expressions to be obtained where the Reynolds number is independent either from the Euler number, the product between the fractional pressure drop and the inverse square of the Mach number, or from the friction factor, that is to say:

$$\begin{aligned} N_{eu} &= \left(\frac{a}{N_{re}} + b\right) \frac{L_e}{R_{he}} \leftrightarrow \frac{\Delta p}{p} \\ &= \frac{1}{2} N_{ma}^2 \left(\frac{a}{N_{re}} + b\right) \frac{L_e}{R_{he}} \leftrightarrow C_f = \frac{a}{N_{re}} + b \end{aligned}$$

In short, both Hagen-Poiseuille's and Forchheimer's laws are models of incompressible flows in which  $\Delta p/p$  is an explicit function of the squared Mach number and  $C_f$  does not depend on  $N_{ma}$ .

However, Eqs. (4) and (5) also agree with the experimental measurements carried out by Su [28] for steady, compressible flow through stacked screens, as well as with the analyses for Stirling regenerators introduced by Organ [16,17,27]. This author recommends

that the experimental correlations for regenerators be presented using  $N_{ma}$  as a parameter because it is influential at values as low as  $N_{ma} \approx 0.1$ . Although the compressibility effects are not probable at moderate velocities, Organ's recommendation will be followed in the present paper to show how those effects can be evaluated through the different functional influences of  $N_{ma}$ , as seen for the aforementioned models of incompressible flows. Therefore, the independence of both  $C_f$  and  $N_{st}$  on  $N_{ma}$ , as well as the dependence of  $\Delta p/p$  on  $N_{ma}^2$ , are expected in the non-tubular heater tests.

#### 4.1. Experimental test rig

The application of friction factor and Stanton or Nusselt number steady flow correlations for Stirling engine heat exchangers is open to discussion because the Stirling engine operation implies not fully-developed, bidirectional flow with a variable mass rate.

For example, Gedeon and Wood [29] pointed out that there is no essential difference between stationary and oscillating flow under the conditions used to test a few regenerators. Conversely, Isshiki et al. [30] concluded that friction factors for oscillatory flow are larger than those in steady flow. In the same way, Ibrahim et al. [31] studied the pressure drop through complex geometries under oscillatory flow conditions and found that the experimental results for oscillatory flow were 1.2–1.3 times the predicted values for the steady operation.

Regardless, the consensus is that steady flow tests may be meaningful to characterise Stirling engine heat exchangers. Therefore, an experimental test rig was constructed (see scheme in Fig. 4) to obtain the correlations for the experimental heater based on the steady flow adaptations of Eq. (4) or (5) and (7).

The compressed air used in the tests was pre-processed at the supply network for moisture prevention and filtering. An additional filter was also arranged at the entry of the mass flow meter, which allows the mass flow to be controlled in the range from 0 to 19 kg/h, with variations of 0.02 kg/h.

The absolute air pressure was measured at the heater entry using a transducer controlled in the range of 1–6 bar through two valves, the first one placed at the connection to the compressed air network and the other one, at the heater exit. The pressure drop across the heater was measured using a water column differential manometer that was also controlled by those valves in the range of 0–1200 Pa. The full-scale accuracy is  $\pm 0.25\%$  for the pressure transducer, and  $\pm 0.5\%$  for the pressure drop gauge.

The air temperature was calculated at both the entry and exit heater sections and the average value of four measurements was obtained by type K thermocouples (200  $\mu\text{m}$  wire diameter). The wall temperature was measured as close as possible to the bases of

the heater rods and to the heater centre using other similar thermocouples inserted in a hole. An additional thermocouple provided the feedback signal for the electrical resistance controller used for heating. The thermocouples used were manufactured under the IEC 584-2 (1982) Standard, which implies an accuracy of  $\pm 1.6^\circ\text{C}$  for the temperature range measured.

#### 4.2. Experimental results

The experiments were carried out at heater wall temperatures of  $100^\circ\text{C}$ ,  $200^\circ\text{C}$ ,  $300^\circ\text{C}$  and  $400^\circ\text{C}$ . The gas pressure and mass flow rate were regulated to obtain series of constant  $N_{ma}$  from 0.0010 to 0.0180, which implies values of  $N_{re}$  between 60 and 1400. The temperature ratios oscillated between 0.045 and 0.325 for  $\Delta T_g/T_w$  and between 0.725 and 0.950 for  $\bar{T}_g/T_w$ . Due to variations in the gas properties,  $N_{pr}$  varied in the range from 0.745 to 0.765.

Fig. 5 shows the fractional pressure drop values obtained for 183 experimental points measured in the abovementioned ranges. The following correlation is a particular case of the Eq. (5) for the experimental heater value of  $R_{he}/L_e = 0.0065$  and adjusts to the measurements with an  $R$ -squared value of 0.9985 and an RMSE of 7.81%:

$$\frac{\Delta p}{p} = 174.058 N_{ma}^{2.079} N_{re}^{-0.238} \left( \frac{\bar{T}_g}{\bar{T}_w} \right)^{-0.941} \left( \frac{\Delta T_g}{T_w} \right)^{-0.033} \quad (8)$$

Fig. 5 must be interpreted by first noting that data with the same  $N_{ma}$  symbol, in general, have different values of temperature ratios. Moreover, the lines of constant  $N_{ma}$  were calculated from Eq. (8) for the mean values of the temperature ratios, that is to say, for  $\bar{T}_g/T_w = 0.86$  and  $\Delta T_g/T_w = 0.200$ , to provide some graphical idea of the dispersion caused by these ratios.

Conversely, the experimental  $N_{ma}$  exponent obtained, which was close to 2, and the inspection of Eq. (5), seem to confirm that the compressibility effects are negligible. Additionally, the near-zero exponent of  $\Delta T_g/T_w$  reflects its limited influence on the fractional pressure drop. Thus, taking these pieces of evidence together suggests using the following friction factor correlation, which is simpler than Eq. (8) and has sufficient accuracy in practise because it adjusts to the experimental data with an  $R$ -squared value of 0.9990 and an RMSE of 9.02%:

$$C_f = 0.904 N_{re}^{-0.145} \left( \frac{\bar{T}_g}{\bar{T}_w} \right)^{-0.991} \quad (9)$$

Fig. 6 shows the experimental data and prediction lines of the constant  $\bar{T}_g/T_w$  deduced from this equation.

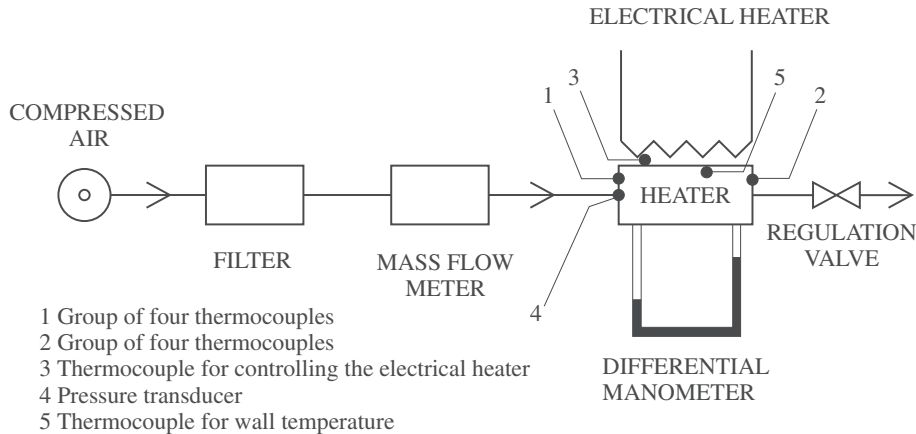


Fig. 4. The schematic of the experimental test rig.

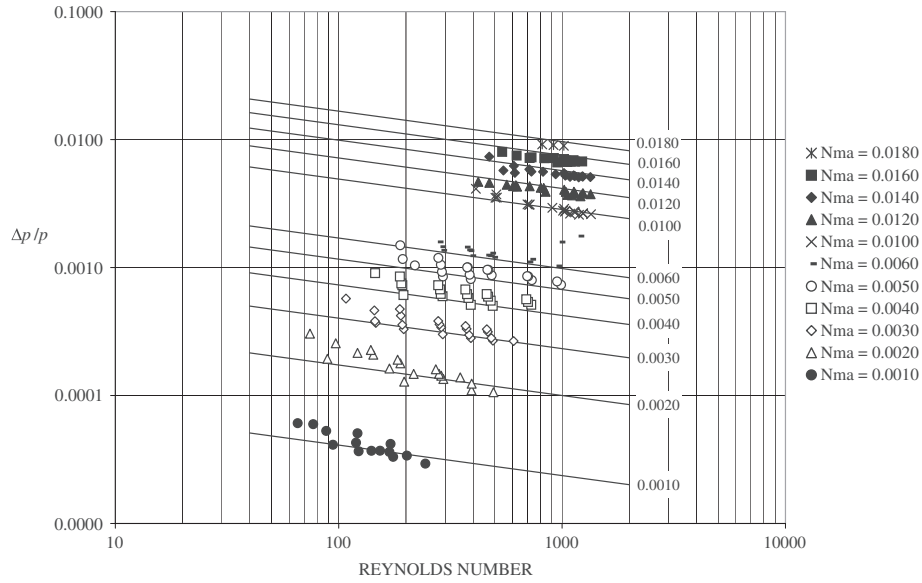


Fig. 5. The fractional pressure drop data obtained from experimental tests.

Figs. 7 and 8 show the Stanton number values obtained for the 183 experimental heat transfer points measured in the above-mentioned ranges. The following correlation is a particular case of Eq. (7) for the experimental heater value of  $R_{he}/L_e = 0.0065$  and adjusts to the measurements with an  $R$ -squared value of 0.9958 and an RMSE of 7.47%:

$$N_{st} = 0.0176 N_{re}^{0.091} N_{ma}^{-0.027} N_{pr}^{1.870} \left( \frac{\Delta T_g}{T_w} \right)^{0.856} \left( \frac{\bar{T}_g}{T_w} \right)^{4.940} \quad (10)$$

The steady flow conditions justify the assumption that the influences of the transient heat conduction variables, that is to say  $N_{TCR}$  and  $N_{fo}$ , were not noticeable.

Fig. 7 shows the lines of constant Mach number deduced from the correlation for the constant values  $\bar{T}_g/T_w = 0.86$ ,  $\Delta T_g/T_w = 0.200$  and  $N_{pr} = 0.75$ . Fig. 8 shows the lines of constant  $\Delta T_g/T_w$  deduced from the correlation for the constant values of  $N_{pr} = 0.75$ ,  $N_{MA} = 0.0075$  and  $\bar{T}_g/T_w = 0.86$ . In this situation,  $N_{st}$  increases

slightly with  $N_{re}$ , while  $N_{ma}$  has little influence, and the dispersion of the results is caused mainly by the temperature ratios  $\Delta T_g/T_w$  and  $\bar{T}_g/T_w$ .

Therefore, as seen in the fractional pressure drop, the compressibility effects associated with the Mach number influence seem negligible, which suggests using the following simpler correlation that adjusts to the experimental data with an  $R$ -squared value of 0.9955 and an RMSE of 7.57%:

$$N_{st} = 0.023 N_{re}^{0.066} N_{pr}^{1.750} \left( \frac{\Delta T_g}{T_w} \right)^{0.856} \left( \frac{\bar{T}_g}{T_w} \right)^{4.975} \quad (11)$$

Additionally, unlike in tubular geometries, in this situation, the Reynolds analogy is not fulfilled.

Conversely, the comparisons with classical correlations show the considerable influence of the temperature ratios in Eqs. (9) and (11). Pending a qualitative explanation that might be derived from further research, these tests have conclusively shown the need to

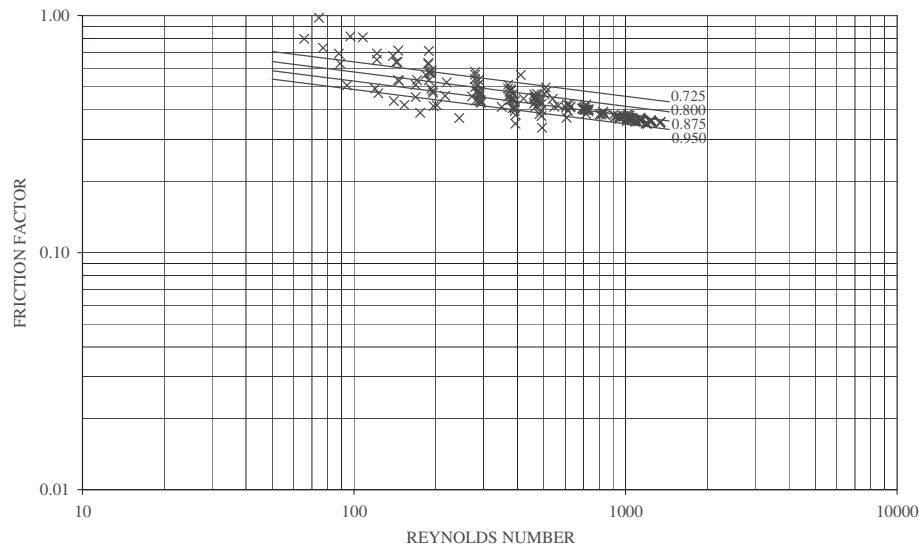
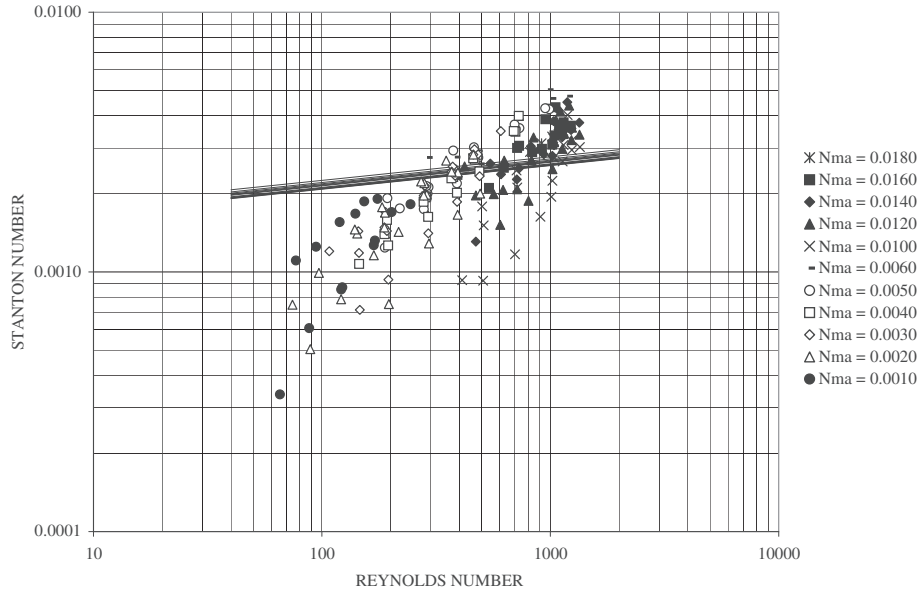


Fig. 6. The comparison between the experimental friction factor and the prediction lines for a constant  $\bar{T}_g/T_w$  deduced from Eq. (9).



**Fig. 7.** The comparison between the experimental Stanton number and the lines of constant Mach number deduced from Eq. (10) for constant values of  $N_{pr} = 0.75$ ,  $\bar{T}_g/\bar{T}_w = 0.86$  and  $\Delta T_g/T_w = 0.200$ .

include the temperature ratios as correlation parameters to obtain accurate results, and the decrease in accuracy if the exponents of those ratios are diminished. Fig. 9 shows that calculations based on the Eq. (11) fit most experimental data within to  $\pm 10\%$  of the dashed lines, while, for example, the following equation adjusts to the measurements with an  $R$ -squared value of 0.9818 and an RMSE of 17.5%, with most experimental data within  $\pm 25\%$ :

$$N_{st} = 0.00086 N_{re}^{0.391} N_{pr}^{1.5} \left( \frac{\Delta T_g}{T_w} \right)^{0.5} \left( \frac{\bar{T}_g}{\bar{T}_w} \right)^{0.5} \quad (12)$$

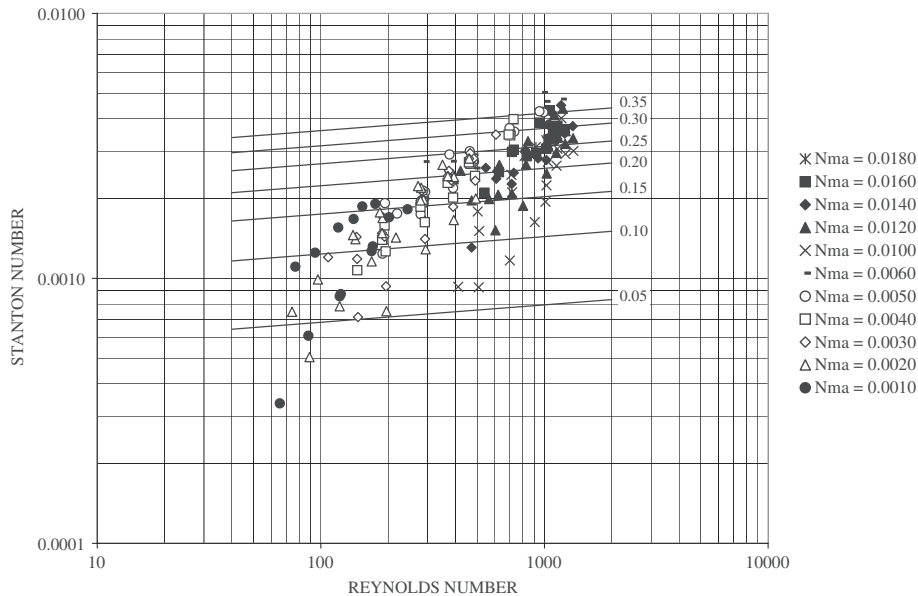
## 5. Discussion

A complete analysis of the predictions of Eqs. (9) and (11) from the viewpoint of the engine operation is outside of the scope of this paper. However, Table 3 provides some approximate idea by

comparing these predictions with the friction factor and the Stanton number values of benchmark prototypes.

Turbulent flow is usually present in Stirling engine heaters [32], and the Reynolds number reaches its maximum values of around 5100 in the M102C engine heater at the operating point assumed for scaling (Table 1), and around 9400 in the heater of the relaxed derivative, due to its out-to-scale, higher swept volume. The rest of the  $N_{re,max}$  values in Table 3 have been estimated from the works of Seume and Simon [32] and Fano [33]. The correlation assumed to calculate the friction factor for slotted or tubular heaters has been recently used by Organ [34] to evaluate the available work lost due to pumping losses in Stirling engine heat exchangers, namely  $C_f = 0.462 N_{re}^{-0.2}$ . The classical correlation by Dittus and Boelter was used for the Stanton number, namely  $N_{st} = 0.023 N_{re}^{-0.2} N_{pr}^{0.667}$ .

The comparisons show that the friction factor values are higher for the experimental heater than for the tubular heaters in other



**Fig. 8.** The comparison between the experimental Stanton number and the lines of constant  $\Delta T_g/T_w$  deduced from Eq. (10) for constant values of  $N_{MA} = 0.0075$ ,  $N_{pr} = 0.75$  and  $\bar{T}_g/\bar{T}_w = 0.86$ .



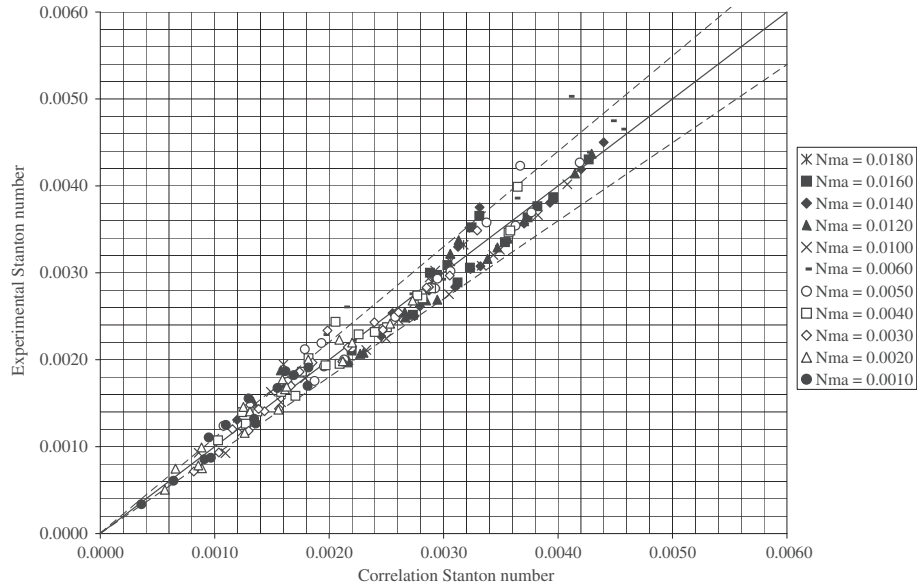


Fig. 9. The comparison between the experimental Stanton number and the correlation estimations based on Eq. (11).

Table 3

A comparison between the friction factors and Stanton numbers for heaters of benchmark engines.

| Prototype          | $N_{re,max}$ | Friction factor, $C_f$ | Stanton number, $N_{st}$ |
|--------------------|--------------|------------------------|--------------------------|
| 4-215              | ~200,000     | 0.04                   | 0.002                    |
| P-40               | ~100,000     | 0.05                   | 0.003                    |
| GPU-3              | ~10,000      | 0.07                   | 0.004                    |
| M102C              | ~5100        | 0.08                   | 0.005                    |
| Relaxed derivative | ~9400        | 0.28                   | 0.003                    |

engines, which probably cancels out the aforementioned advantage of having a favourable  $R_{he}/L_e$  ratio. Conversely, the Stanton number estimations provided by Eq. (11) seem more acceptable, and it seems interesting to note that correlations with lower exponents of temperature ratios, like Eq. (12), would lead to higher  $N_{st}$  values.

In short, from the standpoint of both pressure loss and heat transfer, it seems that a heater with the type of geometry proposed cannot overcome the characteristics of a tubular model. However, it cannot be excluded that an optimised non-tubular heater could be interesting for applications where high Reynolds numbers are reached in the heater and there are practical problems associated with the use of bundles of tubes, given that the available work lost due to heat transfer irreversibilities in tubular heat exchangers usually overwhelms the pumping loss by an order of magnitude [34].

Perhaps the most advisable alternative to optimise the experimental heater geometry is to validate a 3D-CFD model through experimental results, like those provided by this paper. The CFD technique is finding increasing application to the modelling of both Stirling engine components [35,36] as the global engine performance [37]. Such a model could be the basis for analysing the range of performance for which the non-tubular geometry is appealing, taking into account the reciprocating character of the gas flow in the actual engine operation.

## 6. Conclusions

A micro co-generation solar unit has been designed by scaling the Philips M102C beta prototype, assuming a case of relaxed similarity.

The dead volume obtained in the non-tubular approach considered for the derivative heater is more than twice the value of the dead volume in the strictly similar heater. This disadvantage has been compensated for by the reduction of dead volume in the adjacent spaces, which resulted in a computed value  $\zeta_0 \approx 0.302$  for the relaxed derivative, which is slightly higher than the value for the prototype, despite the different engine configuration and drive mechanisms.

The experimental heater ratio  $R_{he}/L_e$  is 41% higher than the strictly similar value, but the dimensionless characteristic parameter of heat exchange  $\alpha_{wxe}$  is 19% lower for the relaxed approach, despite the fact that the wetted surface is almost identical in both cases.

The variables needed to characterise the friction and heat transfer performance of the experimental heater have been identified, appropriate functional relationships have been derived by means of dimensional analysis, and experimental data have been obtained in steady flow conditions.

The  $N_{ma}$  exponent, which is close to 2 in the fractional pressure drop correlation, seems to confirm that the compressibility effects are negligible. Additionally, the near-zero exponent of  $\Delta T_g/T_w$  reflects its limited influence on the fractional pressure drop.

The following friction factor correlation adjusts to the experimental data with an  $R$ -squared value of 0.9990 and an RMSE of 9.02%:

$$C_f = 0.904 N_{re}^{-0.145} \left( \frac{\bar{T}_g}{T_w} \right)^{-0.991} \quad (13)$$

The compressibility effects associated with the Mach number's influence seem also to be negligible in the Stanton number measurements, which suggests using the following correlation that adjusts to the experimental data with an  $R$ -squared value of 0.9955 and an RMSE of 7.57%:

$$N_{st} = 0.023 N_{re}^{0.066} N_{pr}^{1.750} \left( \frac{\Delta T_g}{T_w} \right)^{0.856} \left( \frac{\bar{T}_g}{T_w} \right)^{4.975} \quad (14)$$

The Reynolds analogy is not fulfilled by the experimental heater, and the friction factor values predicted by the Eq. (13) are higher than those estimated for the tubular heaters of benchmark engines,

which probably cancels out the aforementioned advantage of a favourable  $R_{he}/L_e$  ratio. Conversely, the Stanton number estimations seem more favourable.

An optimised non-tubular heater may still be interesting for applications where high Reynolds numbers are reached in the heater and there are practical problems associated with the use of bundles of tubes. An advisable alternative to optimise the experimental heater geometry would be the construction of a 3D-CFD model, validated with abovementioned experimental data.

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